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F1B B2N1 B2N16A B2N16B
F1Q QDD Q216 Q221 Q314 Q511 Q512

(56) Documents cited
GB 2080432 A GB 1252690 A GB 1057361 A
GB 0718100 A GB 0589281 A GB 0476729 A
US 5033269 A US 4445337 A

(58) Field of search
UK CL (Edition K) F1B, F1Q QDD
INT CL[5] F02B, F02C, F02D

(54) **Differential drive for supercharged engine**

(57) A supercharging compressor 6, for an i.c. engine, is driven via a planetary gear train, the planet gear carrier 1, of which is driven by the engine, and the ring gear 3, of which is driven by an exhaust driven turbine 4, via pinion 2. In an alternative arrangement (figure 2) speed increasing gears (8, 9) are interposed between the turbine and the pinion 2 and between the sun gear 5 and the compressor 6.

At low engine speeds the compressor is able to be driven only by the engine at its maximum speed and hence its maximum boost pressure.

The turbine can be used to maintain the compressor at maximum speed and maximum boost pressure, throughout the useful engine operating speed range.

The turbine also provides part of the power to drive the compressor, and augments the power produced by the engine. It is stated that the turbine could be replaced by an electric, hydraulic or pneumatic motor.

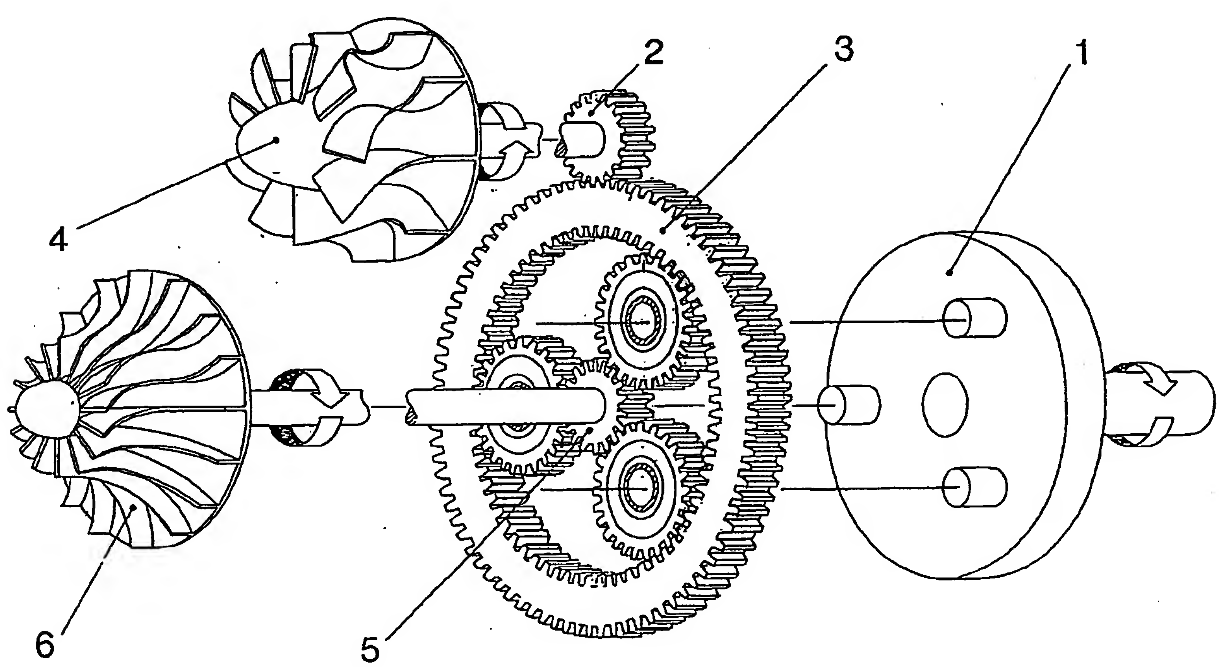

FIGURE 1

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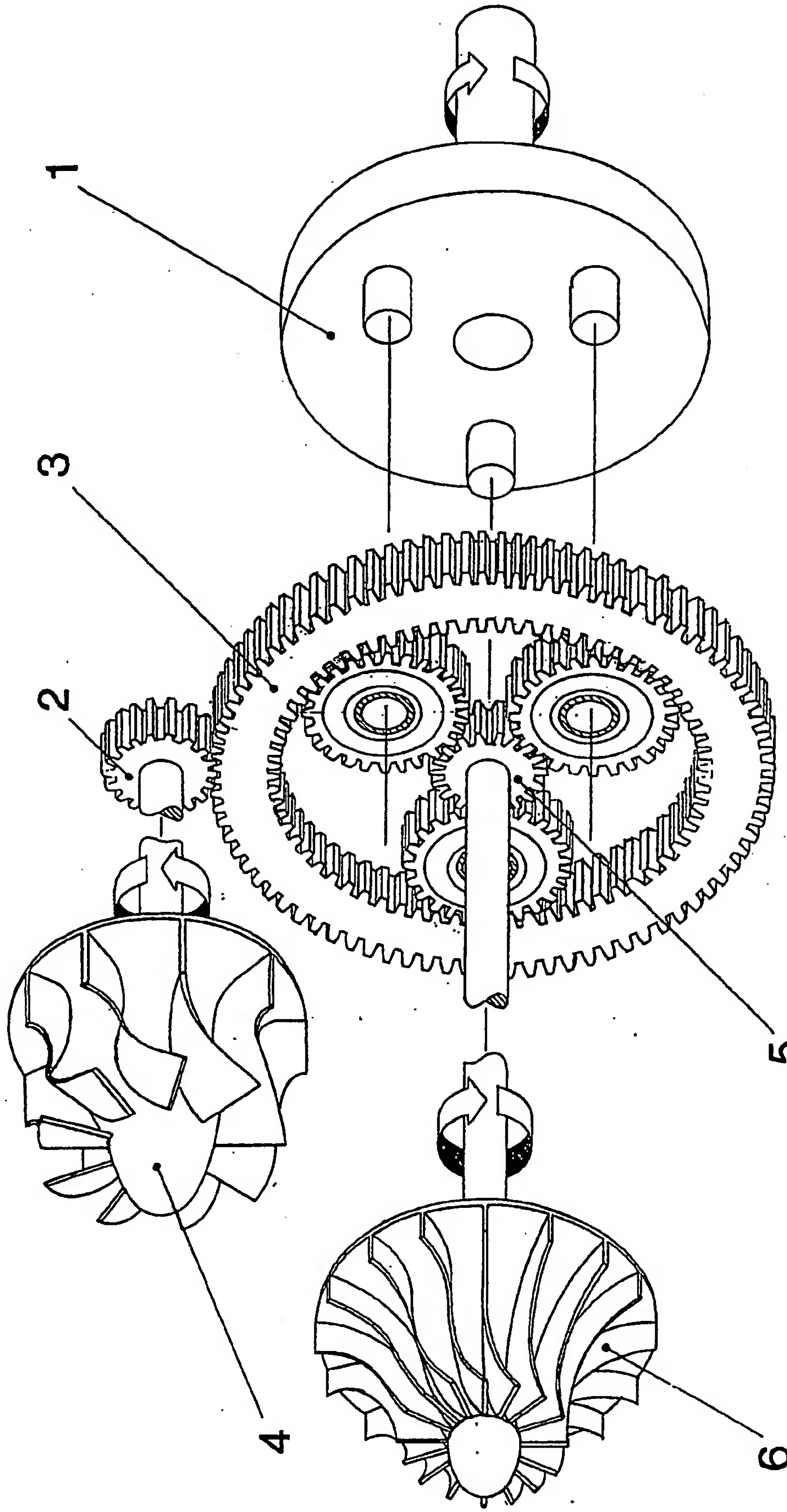


FIGURE 1

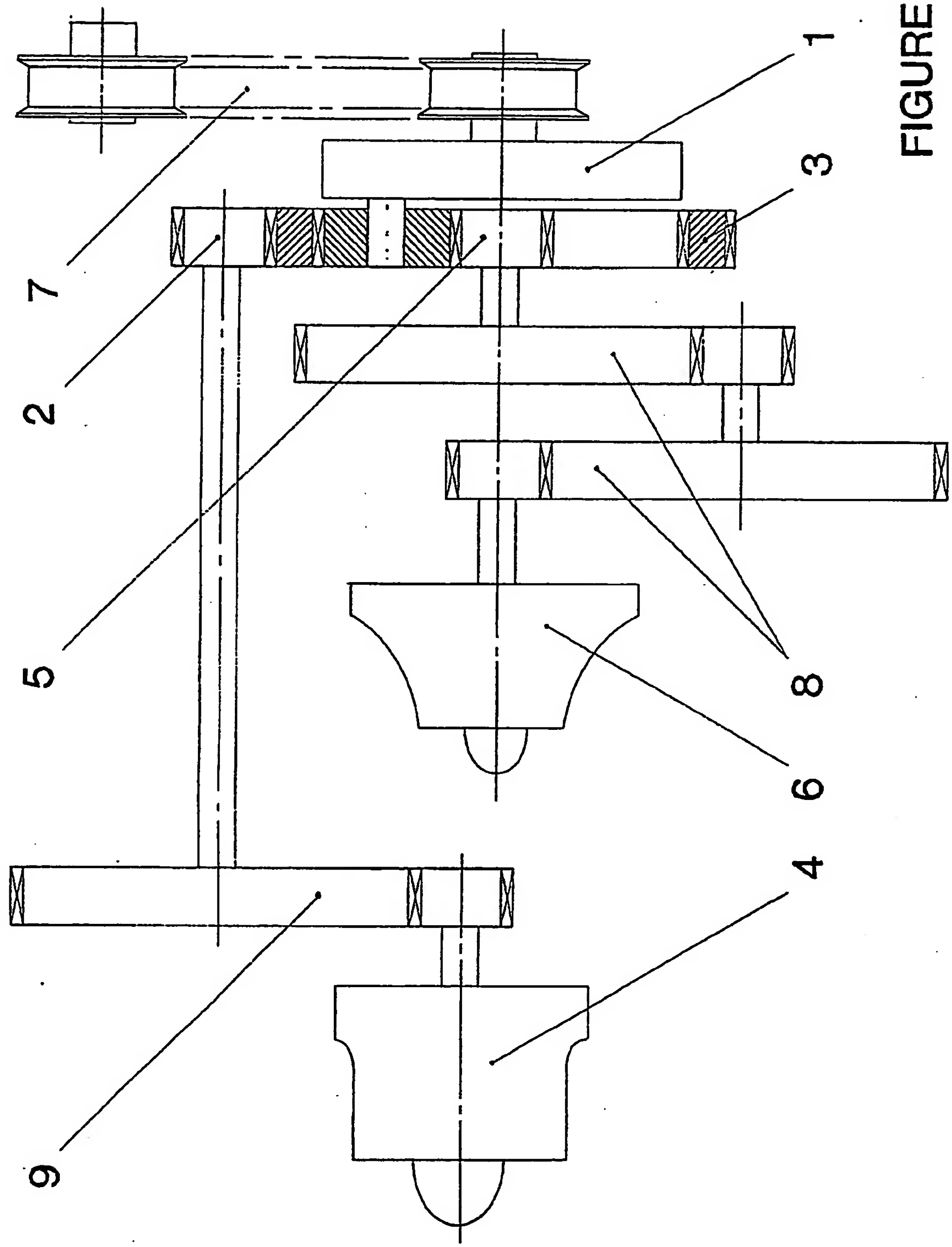
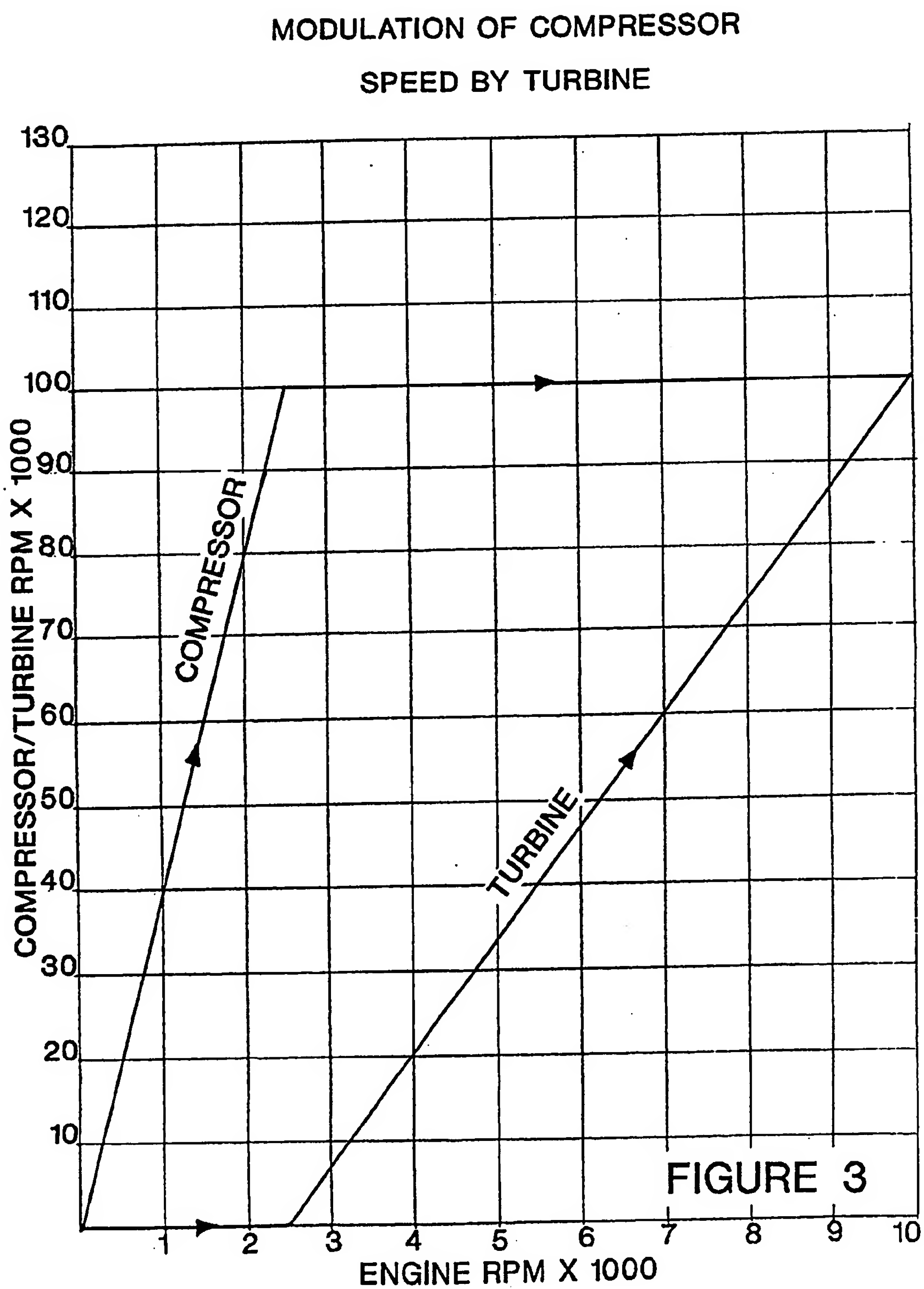


FIGURE 2



MODULATION OF COMPRESSOR
SPEED BY TURBINE
130
120
110
100
90
80
70
60
50
40
30
20
10
COMPRESSOR/TURBINE RPM X 1000
COMPRESSOR
TURBINE
1
2
3
4
5
6
7
8
9
10
ENGINE RPM X 1000

FIGURE 3

MODULATION OF COMPRESSOR

SPEED BY TURBINE

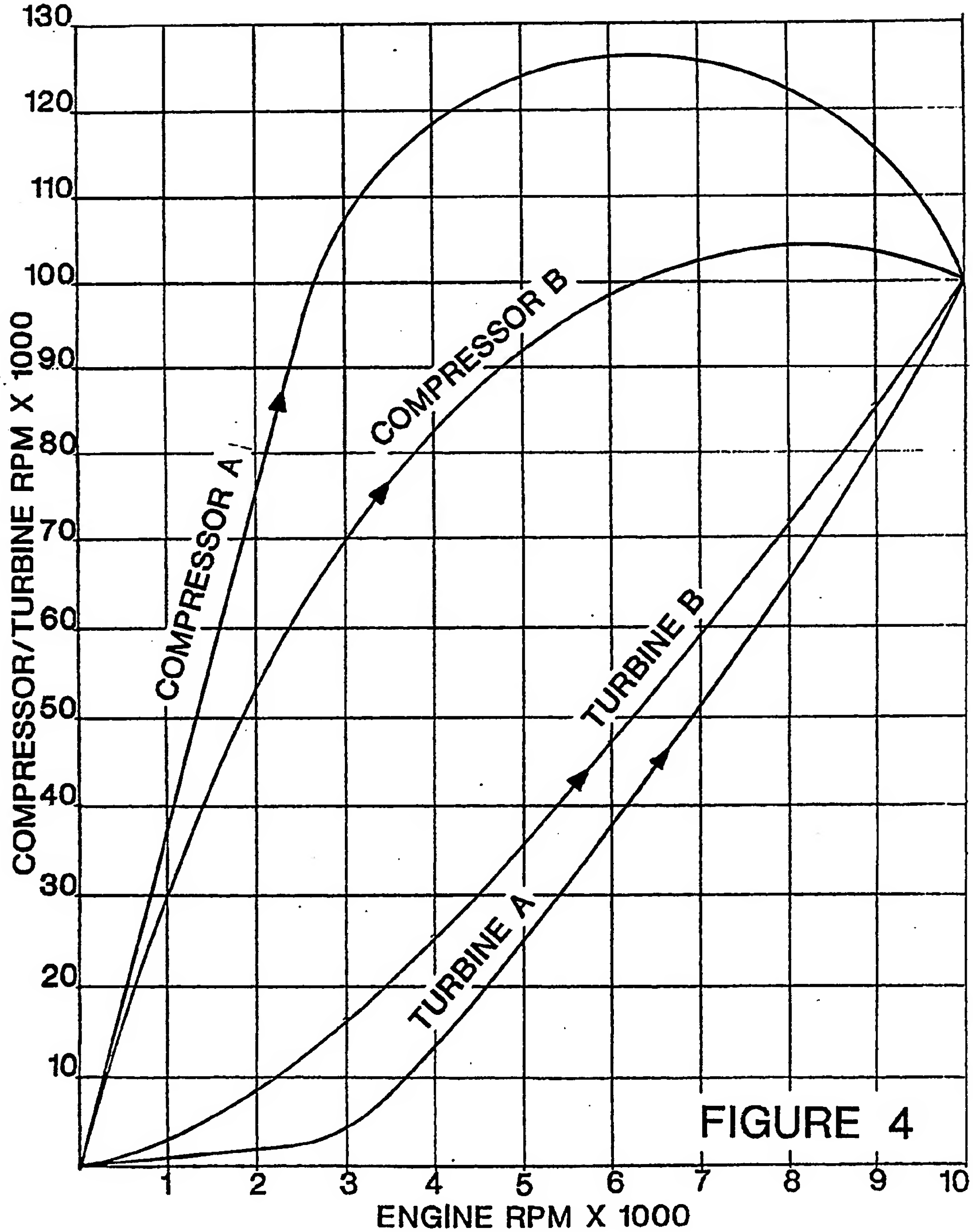


FIGURE 4

DIFFERENTIAL TURBOCHARGER

This invention relates to the supercharging, or charge air compression, of internal combustion engines.

Charge air compression in an internal combustion engine allows the power for a given engine capacity, or swept volume, to be substantially increased.

Displacement compressors of the sliding vane type or rotary lobe type have been much used in the past, but tend to be large and heavy, and of lower efficiency than centrifugal compressors for the same application.

Centrifugal charge air compressors are small and light, and are efficient and effective at full speed. However, when directly geared to an engine, they produce very little boost pressure at low and medium engine speeds, because their operating characteristics are not well matched to the requirements of fixed displacement internal combustion engines.

Turbochargers, in which a centrifugal charge air compressor is driven directly by a centripetal turbine extracting power from the engine exhaust gas, are currently favoured. Turbochargers are not mechanically linked to the engine. This fact allows the compressor to run closer to optimum speed, and thus produce a charge air boost pressure, which is more closely matched to engine requirements.

However, because of the absence of any mechanical link, turbochargers tend to be slow in responding to engine throttle demand. "Turbo lag", as it is known, can be of significant disadvantage. By careful design of the engine system and its components, turbo lag can be minimised but not completely eliminated. Design compromises necessary to minimise turbo lag may adversely affect other aspects of engine performance, such as thermal efficiency, or specific power output.

According to the present invention there is provided a differential drive system for centrifugal charge air compressors for internal combustion engines, in which the compressor speed, and therefore the boost pressure developed by the compressor, can be controlled throughout the useful engine speed range. The system uses an exhaust gas turbine to modulate the effective speed ratio between the engine and the compressor, to give control of the compressor speed.

Modulation of the compressor speed enables the engine torque to be optimised throughout the engine speed range. This offers the possibility of an extremely flexible engine for applications requiring variable speed operation, such as, for example, road or rail transport. The improvement in flexibility might well reduce the complexity and reduce the number of gears required in the main transmission gear box.

The differential system proposed for speed modulation of the compressor offers another benefit, in that surplus power from the exhaust gas turbine which is not required to drive the compressor, augments the engine power. The system can therefore improve the specific fuel consumption, and also the specific power output of the engine.

A specific embodiment of the invention will now be described with reference to :-

Figure 1.	Perspective view
Figure 2.	Gear train arrangement with additional speed increasing gears.
Figure 3.	Example of compressor speed modulation by turbine.
Figure 4.	Further examples of compressor speed modulation by turbine.

Figure 1. shows a perspective view of a differential drive system in which the differential effect is achieved with an epicyclic gear.

In Figure 1. the engine drives the planet gear carrier 1 of the epicyclic gear system. The compressor 6 is driven through the sun wheel 5 of the epicyclic gear system.

At low engine speeds the ring gear 3 of the epicyclic is required to be kept stationary, or only driven at very low speed, so that the compressor is driven primarily by the engine.

By selecting appropriate gear sizes, the sun wheel, or compressor pinion, can achieve its maximum operating speed at a relatively low engine speed.

If the ring gear of the epicyclic gear system is then arranged to rotate at some constant speed, this will change the effective gear ratio between the engine and the compressor.

To facilitate this effect, the ring gear of the epicyclic system is equipped with internal and external teeth, and according to the present invention, an exhaust gas turbine 4 is arranged to drive the ring gear through a pinion 2 meshing with the external teeth. For a system constructed as in Figure 1. rotation of the turbine by the engine exhaust gas will have the effect of slowing down the compressor.

In this way, the effective gear ratio between the engine and the compressor can be modulated in accordance with the ring gear speed, and consequently therefore, in accordance with the turbine speed.

By the above means the boost pressure of the centrifugal charge air compressor, and consequently also the engine torque, can be maximised over the whole of the useful operating speed range of the engine.

Maximum boost pressure for a given engine speed, is defined as the highest boost pressure which the compressor can achieve, at the charge air suction volume flow dictated by the engine speed, whilst avoiding the compressor surge line.

The shaft speeds of the differential gear system shown in Figure 1. are given by the equation :-

$$n_c(D_c) = n_e(D_p + D_c) - n_t(D_t)D_{ri}/D_{ro}$$

Where n_c = compressor pinion speed
n_t = turbine pinion speed
n_e = planet carrier speed

D_c = compressor pinion pitch circle diameter
D_t = turbine pinion pitch circle diameter
D_p = planet wheel pitch circle diameter

D_{ri} = inner ring gear pitch circle diameter
D_{ro} = outer ring gear pitch circle diameter

For vehicle applications there may be a requirement for additional speed increasing gears between the compressor pinion of the differential gear system, and the compressor itself, and also between the turbine pinion of the differential gear system and the turbine itself, in order that the compressor and the turbine can run at suitably high speeds.

Similarly, it may be convenient to drive the planet carrier of the differential gear, through a speed increasing system to assist in achieving this objective.

In this latter case, the speed increasing system may also allow the differential turbocharger unit to be located away from the engine crankshaft, if the drive is via chains or flexible belts for example.

Where additional speed increasing systems are employed, the following equations will apply :-

$$N_c = (r_c)n_c$$

$$N_t = (r_t)n_t$$

$$N_e = n_e/r_e$$

Where N_c = compressor speed
N_t = turbine speed
N_e = engine speed
r_c = speed ratio compressor to compressor pinion
r_t = speed ratio turbine to turbine pinion
r_e = speed ratio engine to planet carrier

The relationship between engine, compressor, and turbine speeds for specific epicyclic gear dimensions is calculated in the following example :-

compressor pinion pitch circle diameter	D_c = 30 mm
turbine pinion pitch circle diameter	D_t = 30 mm
planet wheel pitch circle diameter	D_p = 45 mm
inner ring gear pitch circle diameter	D_{ri} = 120 mm
outer ring gear pitch circle diameter	D_{ro} = 160 mm
compressor speed at maximum engine speed	N_c = 100,000 RPM
turbine speed at maximum engine speed	N_t = 100,000 RPM
maximum engine speed	N_e = 10,000 RPM
speed ratio compressor to compressor pinion	r_c = 16
speed ratio turbine to turbine pinion	r_t = 4
speed ratio engine to planet carrier	r_e = 1

Figure 2. shows the gear train arrangement for the above example in schematic form. The components shown are numbered as in Figure 1. with in addition, the engine to planet carrier drive system 7, the speed increasing gear drive 8 for the compressor, and the speed increasing gear drive 9 for the turbine.

When the shaft speed equation is calculated for the above values of the main parameters it reduces to the following equation :-

$$N_c = 40(N_e) - 3(N_t)$$

For the compressor to reach its maximum speed of 100,000 RPM at an engine speed of 2,500 RPM, and thereafter run constantly at 100,000 RPM, at any engine speed between 2,500 and 10,000 RPM, the necessary turbine speed profile is defined by the above equation.

Figure 3. shows the calculated turbine speed profile in graphical form. The graph shows that the turbine must be held stationary until the engine reaches 2,500 RPM, and thereafter has to increase as a linear function of engine speed.

Figure 4. illustrates two further examples of compressor speed profiles, and shows the turbine speed profiles which would be necessary to generate them. In these examples the compressor profiles are of a smooth continuous form. The turbine speed profiles are therefore also of a smooth continuous form.

Within the practical limits of gear wheel sizes, and allowable gear wheel speeds, and the necessity of avoiding the compressor surge line, any desirable compressor speed profile may be generated, by suitable control of the turbine speed.

Any system required for controlling turbine speed, in order to achieve a desirable compressor speed profile over the useful operating speed range of the engine, is not part of this invention. It may be assumed for example, to be an additional function of an electronic, or computerised engine management system.

The management system would operate in conjunction with, a turbine brake system, and/or a turbine waste gate, and/or a turbine inlet restriction device, such as an inlet gas throttle, or a moveable inlet guide vane system. Alternatively, or additionally, a turbine discharge restriction device such as a discharge gas throttle or a moveable discharge guide vane system, could be used.

An additional function of such a management system could be to avoid compressor surge. That function of such a management system is also not part of this invention but may be assumed to be achieved by the use of a compressor charge air relief valve in the discharge line from the compressor, and/or a moveable discharge guide vane system in the compressor discharge diffusor, and/or a moveable inlet guide vane system at the compressor inlet.

CLAIMS

1 A differential drive system for centrifugal charge air compressors for internal combustion engines, in which the compressor speed, and therefore the boost pressure developed by the compressor, can be controlled throughout the useful engine speed range. The system uses an exhaust gas turbine to modulate the effective speed ratio between the engine and the compressor, to give control of the compressor speed.

2 A device as claimed in Claim 1, wherein the compressor speed and therefore the boost pressure is not controlled solely by the engine, as in a centrifugal charge air compressor driven solely by the engine.

3 A device as claimed in Claim 1, wherein the compressor speed and therefore the boost pressure is not controlled solely by the turbine, as in a turbocharger system having the compressor driven directly by the exhaust gas turbine.

4 A device as claimed in Claim 1 and Claim 3, wherein at low engine speeds, the compressor is able to be driven, and its speed controlled, solely or primarily by the engine, employing a drive system having a very high effective drive speed ratio, which allows the compressor to reach its maximum operating speed when the engine is rotating at relatively low speed.

5 A device as claimed in Claim 1 wherein the exhaust gas turbine is able to provide part of the power to drive the compressor.

6 A device as claimed in Claim 1, Claim 3 and Claim 5, wherein the compressor and exhaust gas turbine operating speeds can be selected independently, with consequent elimination of constraints on the optimisation of the size, weight, and efficiency of the compressor and of the exhaust gas turbine, when compared with a conventional turbocharger system.

7 A device as claimed in Claim 1 and Claim 3, wherein since the compressor boost pressure is maximised throughout the useful engine speed range, there is no delay between engine throttle demand and engine response, as may be the case with turbocharger systems in which the compressor is driven solely by the exhaust gas turbine.

8 A device as claimed in Claim 1 and Claim 5, wherein the exhaust gas turbine can augment the power produced by the engine, since the exhaust gas turbine is linked to the engine crankshaft by the differential drive system.

9 A device as claimed in Claim 1, Claim 5, and Claim 8 wherein if the compressor is partly or fully, externally braked, and/or if the incoming charge air is allowed to fully or partly bypass the compressor, the whole of the useful power developed by the exhaust gas turbine augments the power produced by the engine.

10 A device as claimed in Claim 1, wherein the intermediate speed shafts in the gear trains between engine and compressor, and which can be controlled to run at nominally constant speed throughout the useful engine speed range, may be employed to drive auxiliary engine components. For example, particularly, generators or alternators, but also hydraulic or pneumatic motors, and coolant pumps or cooling fans.

11 A device as claimed in Claim 1, and Claim 5, wherein the exhaust gas turbine could be replaced by a variable speed electric motor, or an hydraulic, pneumatic, or other prime mover if required, without affecting the compressor drive system and the benefits of that system as described in Claims 1, 2, 3, 4, and 7.

12 A device as claimed in Claim 1 and Claim 6, wherein the exhaust gas turbine can be smaller than in an equivalent conventional turbocharger, since the main means of boost pressure control is by turbine speed variation and not by diversion of the exhaust gas around a basically oversized turbine, as in a conventional turbocharger.

13 A device as claimed in Claim 1, Claim 6, Claim 8, and Claim 9, wherein the exhaust gas turbine size can be usefully made larger than in an equivalent conventional turbocharger, thus maximising the amount of exhaust gas energy which can be recovered in the turbine and used to augment the power produced by the engine.

14 A device substantially as described herein with reference to Figures 1 - 4 attached.

Patents Act 1977
Examiner's report to the Comptroller under Section 17 (The Search Report)

Application number
9200911.7

Relevant Technical fields

(i) UK Cl (Edition K) F1B, F1Q (QDD)

(ii) Int CL (Edition 5) F02B, F02C, F02D

Search Examiner

C B VOSPER

Databases (see over)

(i) UK Patent Office

(ii)

Date of Search
24 JUNE 1992

Documents considered relevant following a search in respect of claims 1 to 10. 12 and 13 (when appended to Claim 1 and Claim 14)

Category (see over)	Identity of document and relevant passages		Relevant to claim(s)
X	GB 2080432 A	(SOUTH) - (see whole document (US Equivalent 4452043)	1 to 5 at least
X	GB 1252690	(KICKBUSCH) - (see whole document but note Figure 6 and page 5 lines 30 et seq, also page 6 lines 16 to 34)	1,2,3 and 5 at least
X	GB 1057361	(WALLACE) - (see whole document)	1 to 6 and 8 at least
X	GB 589281	(CORBITT) - (see whole document)	1 to 5 and 8 at least
X	GB 718100	(NAPIER) - (see Figure 1 and page 3 lines 64 et seq. Also see page 2 lines 90 to 112)	1 to 5 and 8 at least
X	GB 476729	(JUNKERS) - (see whole document)	1 to 3,5 and 6 at least
X	US 5033269	(SMITH) - (see whole document)	1 to3 and 5 at least

Category	Identity of document and relevant passages	Relevant to claim(s)

Categories of documents

X: Document indicating lack of novelty or of inventive step.

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Patents Act 1977
Examiner's report to the Comptroller under Section 17 (The Search Report)

Application number
9200911.7

Relevant Technical fields

(i) UK Cl (Edition) Contd. from page 1

(ii) Int CL (Edition)

Databases (see over)

(i) UK Patent Office

(ii)

Search Examiner
C B VOSPER

Date of Search
24 JUNE 1992

Documents considered relevant following a search in respect of claims

Category (see over)	Identity of document and relevant passages	Relevant to claim(s)
X	US 4445337 (GENERAL) - (note Column 4 lines 62 et seq)	1 to 3 and 5 at least

Category	Identity of document and relevant passages	Relevant to claim(s)

Categories of documents

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